LABYRINTH SEAL THERMOMECHANICAL EFFECTS/GUIDELINES FOR SUCCESSFUL OPERATION

Joe Kulesa
General Electric Co.
Cincinnati, Ohio

You've heard a lot of talk yesterday and today about flows and advanced designs. What I'm going to talk about is the workhorse of seal designs. For the most part, the current production preferred seal design is the labyrinth seal. We have a lot of experience with those seals so I'm going to talk about the labyrinth seal. There are two things in general I want to talk about. One is just some thermal mechanical affects and how that affects the seal rub depth which affects temperatures, radial deflections and flows. And the other thing is general guidelines for successful operation of these labyrinth seals.

Relative to the labyrinth seal's thermal mechanical affect, the operating clearance is really what is important. That's what regulates the flow when we want to maintain the flow. Obviously you need to know what your cold clearance is, and you need to know what the mechanical and thermal radial deflections are both for your rotor and stator. In this case the stator would be a honeycomb seal. And we will focus on these radial deflections and clearances, but you also need to understand here is radial deflection of this rotating seal due to the heat input from the rub. And that may or may not be significant. There's different kinds of seal supports. Some seals are rigidly supported by a disc web and bore. We use a lot of these seals because they help hold the stress down in the seal teeth. And generally speaking, that rigidly supported seal has insignificant radial deflection when you would take a rub on that seal. Now, the opposite is true for flexible seals. A lot of times we like to minimize weight and we have a flexible seal that looks like a cylinder with labyrinth seal teeth on it. It would be lighter, would thermally match the transient, heat transfer rates of the thin stator part that it rubs into. So there's an advantage to going with that kind of design. But it's thin enough such that when you take a rub it's flexible enough that you get a non axi-symmetric radial deflection due to the rub. And that's important if you really want to make sure you understand what your clearances are. So I'm going to show you that these non axi-symetric radial deflections due to the thermals from the heat generated during a rub cause deeper rubs and increase the seal operating clearance and the flow. So as long as you manage that, that's fine, but you have to be aware of this deflection during the rub. There was a non axi-symetric 3D finite element analysis done for a seal. This was an itteritive heat transfer and elastic plastic stress analysis of a rotating seal. The analysis assumed that the initial rub was over 180° circumferential arc length and there was heat input into the seal teeth over this 180° arc length due to the rub. That temperature distribution caused the seal to take on a particular radial deflection distribution which is non axi-symmetric. And as time went on, that deflection pattern caused the rub arc length to decrease so the heat input came in over a smaller arc length and eventually after four seconds, the rub cleared itself. Due to symmetry we only modeled 180° of the seal. The results of the temperature versus circumferential length are shown for the half the circumferential length of the seal. And when we first started out there was no rub, you were at a constant temperature and then on the first time point we had a 90° arc length rub, it's really 180° but half of it would be a 90 due to the symmetry of the model. So for the very first moment in time we're rubbing and putting a significant amount of heat over 90°. And then as we iterated we saw that for a very short period of time, a fraction of a second the radial deflection became non axi-symetric. I'll show you the results of deflections in a minute. The seal tooth would deflect in such a way that the arc length of the rub now reduced down from 90° to some lesser value and the temperature continued to rise. And as time went on the circumferential arc length over which it rubbed kept decreasing and the temperature went up, but finally at the last time point it was rubbing only over a very small arc length. These are all relative deflections as if there was no rub, so it's relative to a non-rub condition. So at the very first time point we just start to pick up a little bit of radial deflection over the 90° arc. And as time increases the radial deflection increases but it's over a very small circumferential arc length. At that last time point the radial deflection is significant and it's over a small arc length and then with the following time point the rub cleared itself and then you were back to a no rub condition. So this was all done from an assumed 180° rub on a rotating seal. With some of the things we've learned if we were to do the analysis again we would have started off with a much smaller initial arc length rub, but none the less it does show that local rubbing produces local heat input and it deflects the seal and it basically clears itself, but it does cause additional radial deflections on this flexible seal that would not have been taken into account with an axi-symmetric analysis. We did also do some seal rub rig component testing and this was done on the same seal shown here. And what we did was measure the tooth tip temperatures during the rub into the honeycomb and we used a pyrometer to measure these temperatures. The pyrometer was looking around the entire circumference of the tooth transiently. The test results show that this tooth tip does produce very localized rubs and gives us this non axi-symmetrical temperature gradient due to the rub. The results of the test shows tooth tip temperature versus circumferential position. This wasn't plotted for all 360° because these are very localized rubs. So you could see at the seven seconds time point, there was no rub and at least no rub according to pyrometer threshold temperature of 1200F. And basically the rub came in around the seven and a half second time point and the temperature built up close to 2000F degrees within just a couple of seconds and then it cleared itself. The arc length of the rub was a very localized 10 to 15°arc length rub. When you look at your hardware you might see that this is in disagreement. The hardware looks like it has rubbed 360° around the circumference, but this component test is showing a very localized rub. But what really happens is during one particular rub event you'll get a localized rub and you wear a spot on that seal and then the next time you would rub it would rub adjacent to either side so as you continue to take additional rubs it's going to rub the entire circumference at one time or another. So in conclusion if you have flexible seals they will rub locally around the circumference of the seal tooth and that causes the non axi-symmetric temperatures and non axi-symmetric radial deflections. And that will give you an increase in honeycomb rub depth and which will give you an increase in clearance and increase in flows. You just have to make sure you take the additional clearance into account.

The next thing I want to talk about, many of you may know a lot about, is summary of the guidelines for successful operation of labyrinth seals. I'm not going to go into this in great detail because I could spend a lot of time talking about the details of these requirements, but these are general requirements. I have listed some references if anyone is interested. In general to prevent vibration there's a lot of modes vibration we can get into. One of them would be just a major Campbell resonance or a major resonance for a rotating seal. So in that situation what you'd really want to do is just avoid having the backward traveling wave frequency of that seal relative to stationary observer become zero. You could also get into a situation where you have a major resonance with a static seal, the honeycomb seal that you rub into or the structure supporting the seal. And what you need to do there is avoid the natural frequency of the static part being equal to the nodal diameter times the number of circumferential nodal diameters of the static seal times the rotor speed, and there's also a reference listed for that. Relative to resident interaction you need to avoid having the rotating and stationary seal having coincident frequencies at the same number of circumferential nodal diameters for both the rotor and the stator. There's also acoustic coupling. What you want to do is avoid coincident frequencies between the mechanical frequency of the rotating seal, and the acoustical cavity down stream of the seal. And finally there's aeroelastic instability that gets tied to the tooth pocket frequency. The tooth pocket frequency is the little cavity just in between the seal teeth themselves. And what you want to do is avoid a tooth pocket frequency relative to mechanical frequency of the seal. And there are certain guidelines that is shown on the next pages. What's important for this aeroelastic instability is the relationship between the acoustic frequency of the little cavity between the seal teeth and the actual mechanical frequency of the seal itself. And that relationship of those frequencies can either act to stabilize any vibration or they can act to destabilize which could cause an aeroelastic instability. So what you want to avoid is this negative damping or instability that could occur under the following conditions. You have a seal that's supported on the high pressure side, if the air was flowing in this case from the left to right. If the seal is supported over here on the high pressure side then what you want to do is have a mechanical frequency less than the acoustical frequency. The plot of aeroelastic damping shown on the Y axis gives you the positive or negative number and obviously you want positive damping not negative damping. The X axis is mechanical frequency of the seal, so if you assume you have a fixed acoustical frequency and you vary your mechanical frequency, and if your mechanical frequency was above the acoustic frequency you'd get the negative damping, and if your mechanical frequency was less than the acoustic you get positive damping. So in this particular situation a negative damping could exist with mechanical frequency being greater than the acoustical frequency. So that's a situation that you want to avoid if it's supported on a high pressure side. Now then the opposite is true if you support it on the low pressure side and the air is still flowing from left to right. If the mechanical frequency was less than the acoustical frequency you can have a negative damping and that would give you the aeroelastic instability. That's really all I had.

Questions

Q. Your rub analysis gets a correlation that's used for rub material heat generation as a function of the rubbed material being lost. To clear the rub you assume some relation between heat generation and the rotor to stator radial interference ?

A. In that particular analysis, it was an analysis where we assumed an initial 5 mil radial interference, as if the rotor had changed speed causing the 5 mil radial interference of the seal. It was just to show that thermally the heat would be generated and you would get a temperature distribution and a radial deflection that would eventually clear itself. We did not consider the actual rubbing out on the material itself. We neglected that.

Q. There's one part stationary and one part moving and you have to go from 0-360° and boundary condition moving circumferentially relative to the rotating seal.

A. That is correct. I guess we really didn't take it to that level of sophistication. I think what they did is they considered the reference frame rotating with the seal. They assumed some condition of rub based on the experiments and then they just tracked the local heat input.

Q. The rubbing generates heat and heats up the air moving circumferentially around the seal. The circumferential air temperature variation would cause temperature gradients for the 360^0 for the seal, but you only considered 180^0 of the seal

A. But your worried about the air temperature variation around the circumference and I think that's a relatively small impact relative to the rate at which the heat is coming in from the rub. In a fraction of a second to tooth tip gets up over 2000F due to the rub. The average air temperature is still moderately cool with a small circumferential temperature variation. But that would be the next level of sophistication, you are right.

Q. What is the change in the design to vary the mechanical and acoustical frequencies?

A. What could we change in the design? Well, you know the acoustical frequencies is simply a function of air temperature and the radius. And normally it's hard to change that. So pretty much you'd have to change the stiffness of the rotating seal by adding or taking away stiffness. That'd be the easiest thing to change.

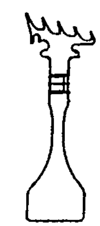
Q. You could also add structural damping through honeycomb or some other device.

A. Yes. You could add external mechanical damping to the seal, you are right.

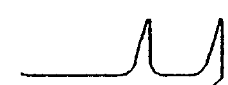
Table of Contents

Labyrinth Seal Thermomechanical Effects

- Seal operating clearance is a function of
  - Cold clearance
  - Mechanical and thermal radial deflection of rotating seal and honeycomb seal
  - Thermal radial deflection of rotating seal due to heat input from rub
    - For seal rigidly supported by disk web and bore radial deflections are insignificant



    - For flexible seals (cylindrical) non axi-symmetric radial deflections can be significant

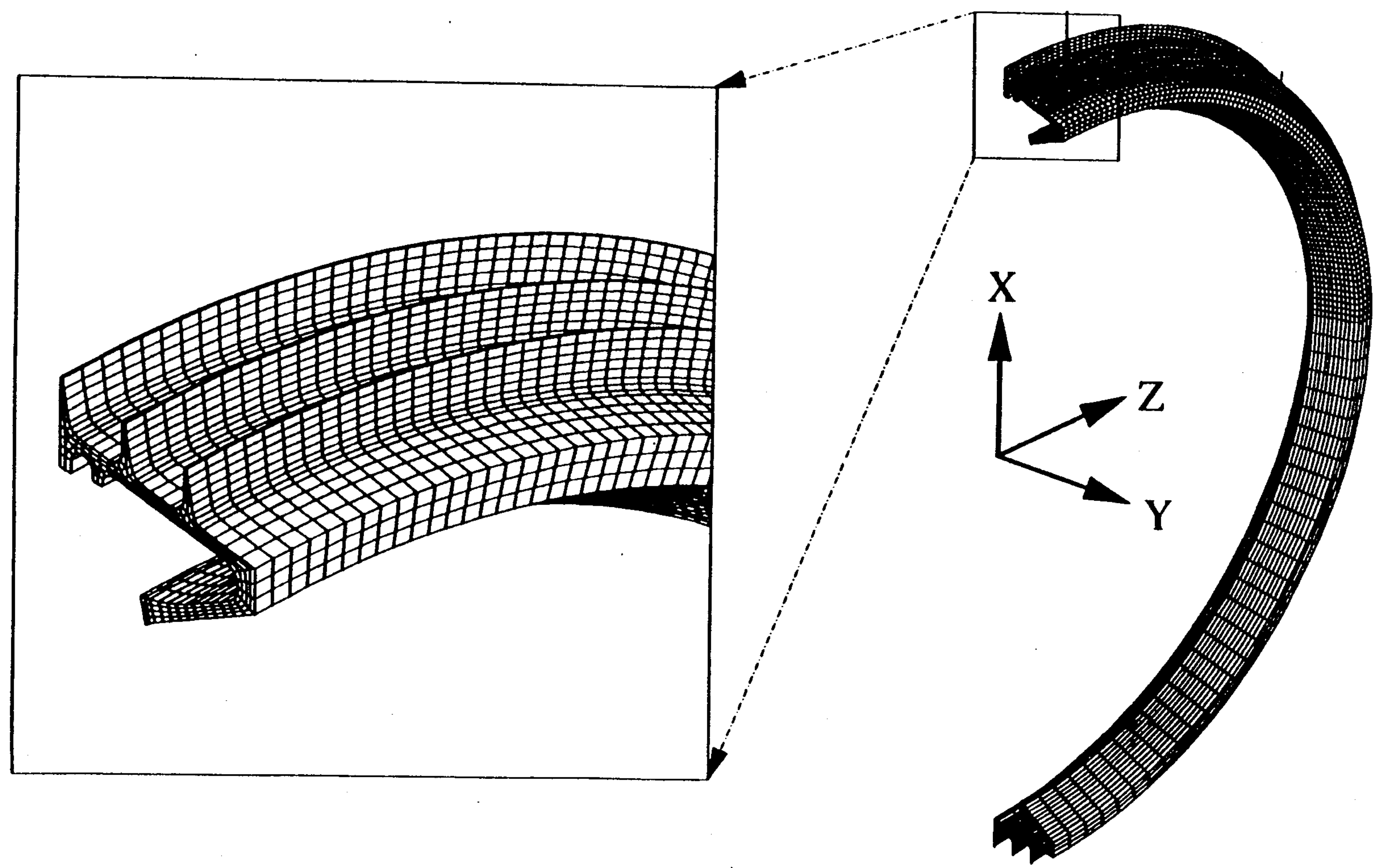


- Non axi-symmetric radial deflection due to thermals from rub can cause deeper rubs and increased seal operating clearances and flows

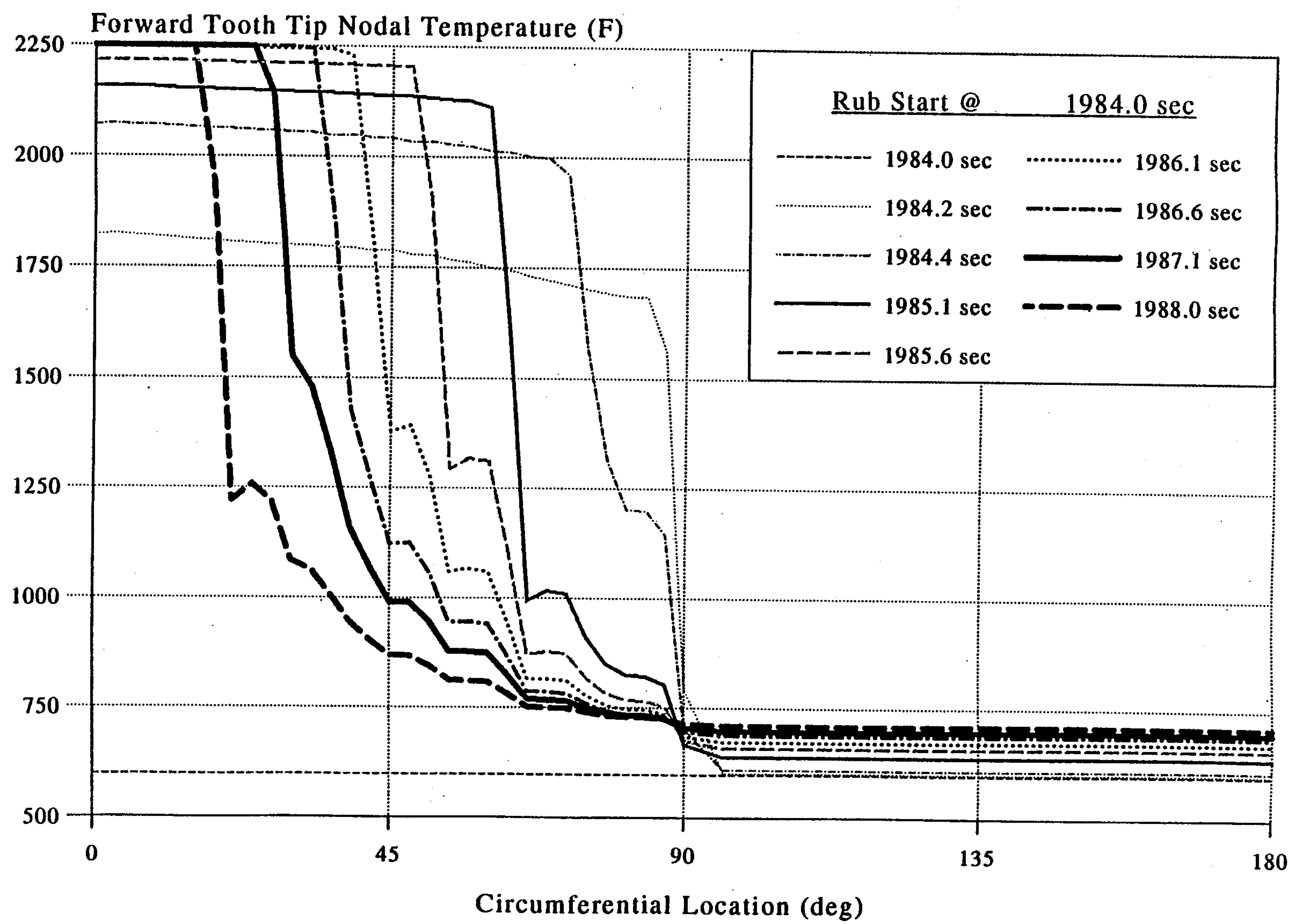
Non Axi-symmetric Seal Radial Deflections

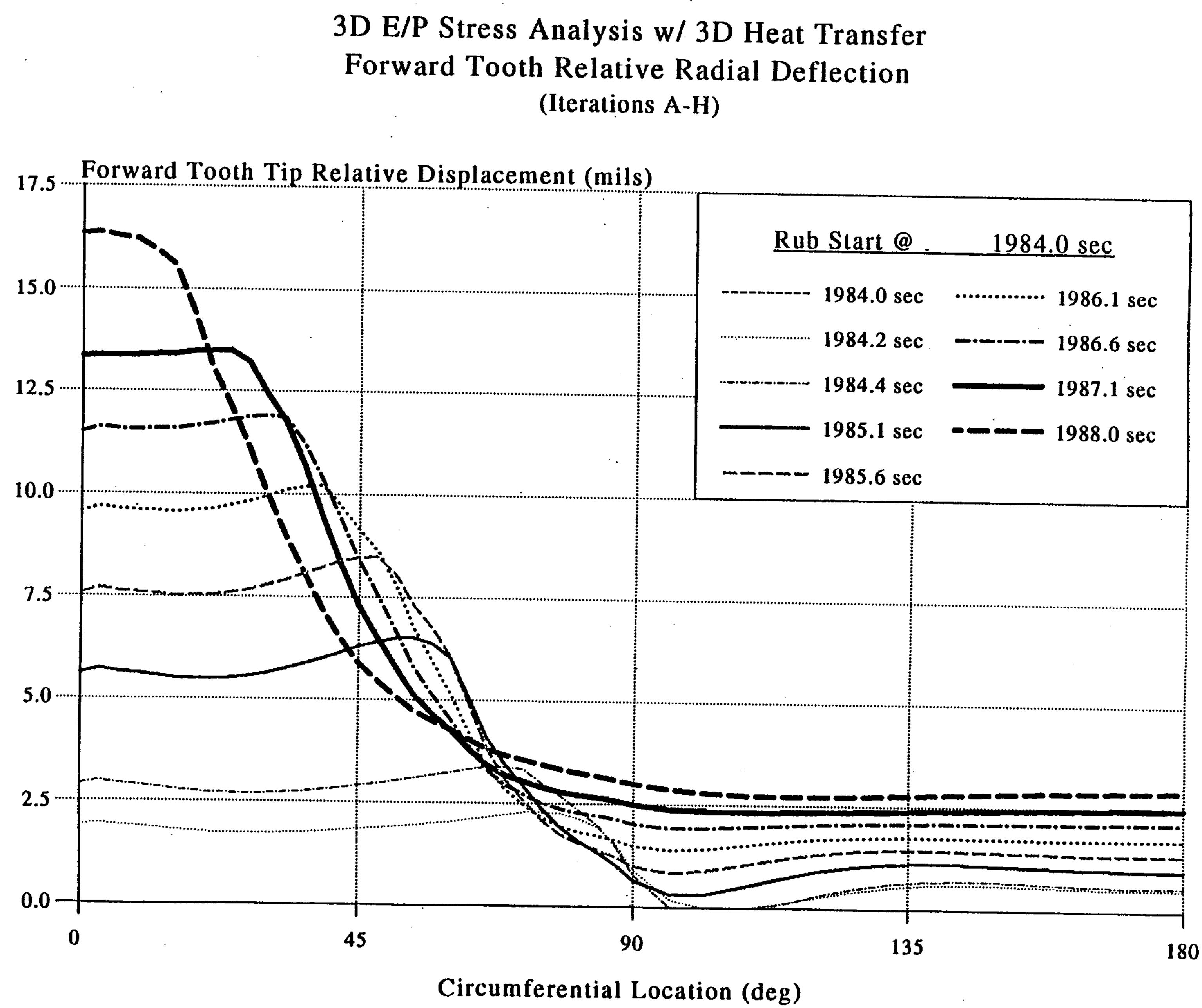
- Iterative elastic/plastic heat transfer and stress analysis of a seal was completed
- Analysis assumed initial rub over a 180 degree circumferential arc length of seal teeth
  - Heat input into seal teeth over the 180 degree rub arc length due to rubbing
  - This temperature distribution caused the seal to have non axi-symmetric radial deflections which caused the rub arc length of 180 degrees to decrease
  - After 4 seconds, the rub arc length reduced to zero causing the rub to clear

3D Elastic/Plastic Stress Analysis of Labyrinth Seal



3D E/P Stress Analysis w/ 3D Heat Transfer
Forward Tooth Tip Circumferential Temperature Variation
(Iterations A-H)

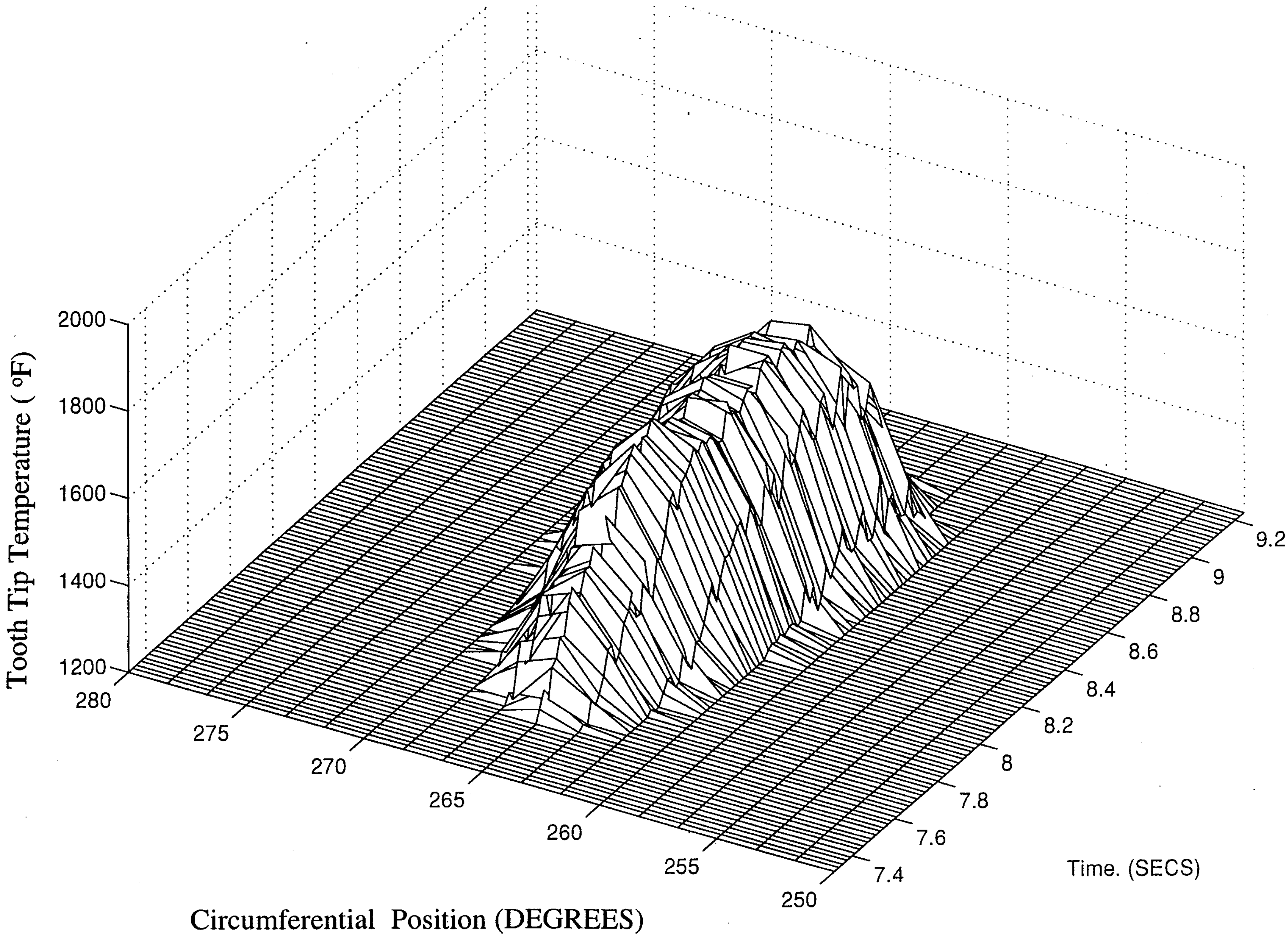



3D E/P Stress Analysis w/ 3D Heat Transfer
Forward Tooth Relative Radial Deflection
(Iterations A-H)
Forward Tooth Tip Relative Displacement (mils)
17.5
15.0
12.5
10.0
7.5
5.0
2.5
0.0
0
45
90
135
180
Circumferential Location (deg)
Rub Start @ 1984.0 sec
1984.0 sec
1984.2 sec
1984.4 sec
1985.1 sec
1985.6 sec
1986.1 sec
1986.6 sec
1987.1 sec
1988.0 sec

Seal Rub Rig Component Test

- Component rub rig of a flexible seal measured seal tooth tip temperatures during a rub into honeycomb using a pyrometer
- Test results show a tooth tip non axi-symmetric temperature due to the local rub
- This verified the analytical results

Labyrinth Seal Rub Rig Test Results



Conclusion

- Flexible seals can rub locally around the circumference of the seal tooth causing:
  - Non axi-symmetric temperatures and radial deflection
  - Increase in the honeycomb rub depths, operating clearance, and flow

Guidelines for Successful Operation of Labyrinth Seals in Jet Engine Environment

Vibration Mechanism	Guidelines
Campbell/Major Resonance for rotating seal	For rotating seal avoid a zero backward traveling wave frequency relative to a stationary observer (Reference 1)
Campbell/Major Resonance for static seal	Avoid frequency of static part= nodal diameter x rotor speed (Reference 2)
Resonant interaction	Avoid coincident frequencies & mode shape between the rotor and stator

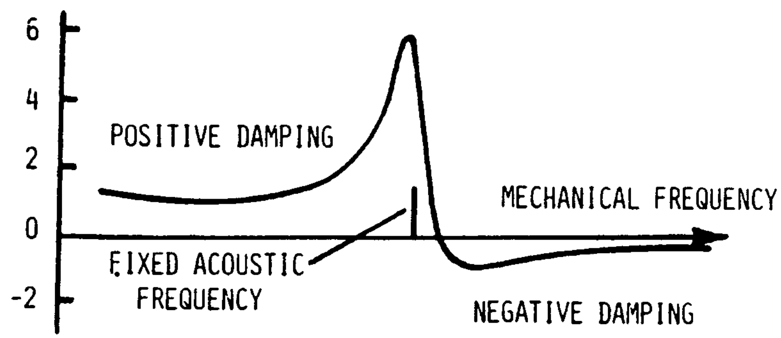
Guidelines for Successful Operation of Labyrinth Seals in Jet Engine Environment

Vibration Mechanism	Guidelines
• Acoustic coupling	• Avoid coincident frequencies between the mechanical and acoustic cavity downstream of the seal (Reference 3)
• Aeroelastic instability with tooth pocket frequency	• Avoid tooth pocket acoustic frequency relative to mechanical frequency as shown on next page (Reference 4)

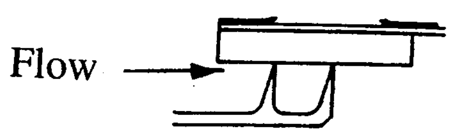
Aeroelastic Instability

- The relationship between the tooth pocket acoustic frequency to the mechanical frequency impacts the ability to stabilize or destabilize the aeroelastic vibration
- Negative damping exists and instability could occur for the following conditions:

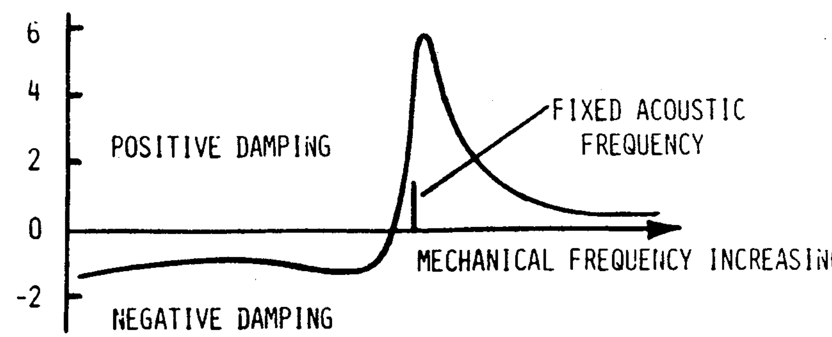
$f_{mech} > f_{acoustic}$



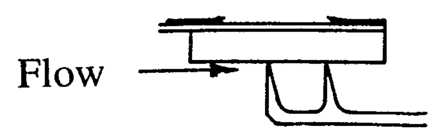
HIGH PRESSURE SIDE SUPPORT



$f_{mech} < f_{acoustic}$



LOW PRESSURE SIDE SUPPORT



References

1. Campbell, W., "The Protection of Steam Turbine Disc Wheels from Axial Vibrations", Trans ASME Vol. 46, 1924.
2. Alford, JS, "Protection of Labyrinth Seals from Flexural Vibration", ASME P63-AHGT-9 Journal of Engineering for Power 1963.
3. Alford, JS, "Protecting Turomachinery from Unstable and Oscillatory Flows", ASME Paper No. 66-WA/GT-13.
4. Abbott, D.R., "Advances in Labyrinth Seal Aeroelastic Instability Prediction & Prevention" ASME 80-GT-151.